# ENERGY INNOVATIONS SMALL GRANT (EISG) PROGRAM

## EISG FINAL REPORT

### ENERGY-EFFICIENT AIR-HANDLING CONTROLS


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## Abstract

We developed a new strategy for efficiently operating air-handling equipment. The strategy takes advantage of the fact that there are usually more control degrees of freedom than control objectives. We used a computer-based model of air-handling system operation to develop the new strategy. The model predicts the transient response of pressures, velocities, and temperatures in the system resulting from the operation of fan and damper controls. The model was calibrated using data from an office building in Oakland, California. The new strategy, called Efficient Air-Handling StrategY (EASY) involves sequential modes of operation that can be represented with a finite state machine. We compared EASY with three existing strategies: two volume-matching strategies and plenum pressure control. The results demonstrate that EASY uses less energy than any of these strategies. The results also demonstrate that resetting supply duct static pressure has a significantly bigger impact on energy performance than efficient operation of the air-handling unit control dampers and return fan. We found that resetting based on critical damper position was the most difficult control loop to tune. The settling time for resetting based on critical damper position was approximately 2 hours. We found that resetting based on supply duct flow provides nearly optimal performance while eliminating problems with reliability and reducing the settling time to 30 minutes.

# Executive Summary

1. Introduction

A large fraction of the energy consumed in buildings is used to operate fans that move air for heating, ventilating, and air-conditioning (HVAC). Much of this fan energy is wasted because fans are not operated as efficiently as possible. Inefficient operation of fans can increase cooling energy consumption in addition to fan energy consumption because inefficient fan operation causes more hot air than necessary to be cooled. In addition to wasted energy, inefficient operation of HVAC fans disproportionately increases peak demand because these systems use the most energy in the afternoon when energy is in shortest supply.

2. Project Objectives

The specific objectives of the project were as follows:

a. develop and demonstrate the benefit of a new, energy-efficient method of controlling air-handling equipment in buildings using optimization and computer simulation methods,
b. reduce fan power by 20%-40%,
c. reduce cooling power by 10%, and
d. ensure that the controls are stable, and that minimum ventilation, appropriate building pressurization, and appropriate temperature control are achieved at all times.

We used an existing model of air-handling systems to achieve the first objective. We calibrated the model using data from an air-handling system in the Elihu Harris State Office Building in Oakland, California. We proposed to use optimization to develop the new strategy. The concept was to first determine optimal system behavior, then identify patterns of behavior that could be encoded as control rules that would achieve nearly optimal performance.

We expected the reductions in energy consumption associated with the second objective to arise from new control concepts designed to reduce throttling losses at the control dampers (exhaust, return, and outdoor air dampers) of the air-handling unit. These devices are all upstream of the supply fan, and their optimal operation has not been studied before. The magnitude of the gains were derived from published research on the energy conservation gains achievable from optimizing the supply fan controls, which operate on sensor signals downstream of the supply fan.

We expected the gains associated with the third objective to arise from better control over outdoor airflow rates. Existing systems that use fixed minimum damper positions for regulating outdoor airflow can over-ventilate when it is hot, increasing cooling power. The proposed strategy directly controls outdoor airflow rate, so cooling power should be minimized.

3. Project Outcomes

We invented a simpler way to model the behavior of fans that significantly reduces the complexity of predicting the transient behavior of fan controls. The existing method requires logic necessary to trap conditions where the fan speed is zero, otherwise the model will divide by zero. Fan speed of zero is a very common operating point for real fan systems. All predictions of start-up and shut-down transients must be able to predict operation at or near a fan speed of zero. The simpler fan component model only needs to trap the case where the fan speed and flow are simultaneously zero. The predicted behavior under this condition is simple: no pressure developed and no power consumed.

We could not get the optimizer to converge to a globally optimal operating point. The problem is that the cost function, total power consumption, has many local minima. The optimizer would move downhill on the cost function and stop at a depression in the cost function that was not the lowest. We proved this by demonstrating the optimizer would converge to operating points that used more power than existing strategies operation under the same conditions.

Since we could not use optimization as intended for design of the new strategy, we used trial-and-error and the behavior of the existing strategies instead. Using these methods, we learned the following facts that guided the design of the new strategy:

a. Reducing static pressure at part-load conditions has the biggest impact on energy performance.

b. Static pressure , when reset based on critical damper position, is highly correlated with supply flow rate.

c. Sequential operation of the dampers reduces fan power.

d. High building pressure reduces fan power when the economizer is enabled.

e. When the economizer is disabled, opening the outdoor air damper and return damper and closing the exhaust damper reduces fan power.

These findings were incorporated into the design of a control system that can be implemented with a finite state machine, where the states are modes of operation such as economizer enabled and economizer disabled. We called this control strategy Efficient Air-handling StrategY (EASY)

We compared EASY to the standard volume-matching strategy, which is used at the Elihu Harris building, a variant of volume-matching that prevents flow reversals when the economizer is disabled, and plenum pressure control (PPC). EASY used less energy than any of the existing strategies, but the improvement is small (2% - 10%). Most of the fan energy reduction resulted from resetting static pressure, not from reducing losses upstream of the supply fan. However, the improvements from this project based on static pressure resetting are new because we reset static pressure based on supply airflow rate instead of the position of the critical terminal damper, which is the standard way to reset static pressure. We call this new resetting method Static Adjustment based on Volume flow (SAV).

Compared to systems without static pressure reset, SAV reduced fan power by 26.3%. SAV also reduced cooling power by 17.4%. The reductions come from reducing fan power and reducing leakage. Reducing fan power reduces cooling power because the fan energy is converted to heat and must be rejected by the cooling system. Lower operating pressures resulting from SAV reduced duct leakage. We found that 25% of the supply air in the air-handling system used to calibrate the model leaked out between the supply fan and the terminal dampers. When it is hot, leakage forces the system to cool extra return air, which is hotter than the supply air. SAV yielded 93% of the benefit of static pressure reset based on the position of the critical damper.

SAV can be applied to any VAV system, including systems with pneumatic controls and legacy DDC systems that cannot report damper position. SAV can be used as a standalone strategy without the other components of EASY. It could be applied to systems with a volume matching strategy or a PPC strategy.

## 4. Conclusions

a. EASY reduces energy consumption by 2% - 10%.

b. EASY with SAV reduces fan power by 26.3% relative to a system without static pressure reset.

c. EASY with SAV reduces cooling power by 17.4% relative to a system without static pressure reset.

d. EASY and SAV are easier to tune that existing strategies, and they have a faster response time when tuned properly.

e. SAV can be implemented in the majority of existing VAV systems that do not have the capability of implementing static pressure reset based on terminal position.

## 5. Recommendations

A field test is needed to quantify and validate the energy efficiency benefits. The tests should be conducted during the summer so that cooling energy benefits and fan energy benefits can be quantified.

Since most of the savings are derived from resetting static pressure, more research should be focused on static pressure resetting. Alternative strategies that can address the problems of resetting based critical damper position should be investigated. Methods for commissioning SAV so that it produces the largest energy benefit should be investigated.

## 6. Public Benefits to California

The project demonstrated that the proposed technology could reduce average power consumption by 0.23 $W/ft^2$/year in buildings with VAV air-handling units. Based on data from DOE, we estimate that the proposed technology, if implemented in all applicable buildings, would reduce energy costs by $928MM/year nationwide and $171MM/year in California.

## Introduction

A large fraction of the fan energy used by HVAC systems is wasted because the HVAC systems are not operated efficiently. Lorenzetti and Norford (1994) showed that fan energy consumption in variable-air-volume (VAV) HVAC systems could be reduced by 19-42% simply by changing the way that the supply fan and control devices downstream of the supply fan are operated. Approximately the same amount of air that flows through the supply fan and control dampers downstream of the supply fan also flows through the return fan and control dampers upstream of the supply fan. However, the operation of the upstream controls have not been optimized because doing so is more complicated than optimizing the behavior of the supply fan and terminal dampers.

The energy required for cooling the air used to ventilate buildings is directly dependent on the outdoor air temperature and humidity. The most common method of controlling the minimum ventilation rate in buildings is to use a fixed minimum position for the outdoor air dampers. Doing so causes the ventilation rate of VAV systems to be highest when the outside air is hottest, which substantially increases the energy required for cooling.

Since the power consumption of air-handling equipment is dependent on weather and the daily timing of internal heating and cooling loads, air-handling equipment contributes disproportionately to problems with insufficient electrical power capacity during hot summer days. These factors mean that improvements in air-handling efficiency would also have a significant impact on the reliability of electric power.

Proper control of air-handling equipment is complicated by three technical problems. The first is that there are multiple, interacting functional requirements that air-handling systems must serve. The functional requirements are to provide the adequate ventilation, appropriate building pressurization, and appropriate temperature control for the building. Ventilation and temperature control requirements are conflicting when it is hot outdoors, but are not conflicting when it is cool outdoors.

The second technical problem is that the controls are strongly coupled. Coupling of the control loops in air-handling units has been documented by Avery (1992) and Kettler (1995). Air-handling units contain a number of final control elements. In larger systems there is either a return fan or an exhaust fan, but not both. Sometimes an injection fan is used to control the minimum ventilation. Larger systems have three or more sets of control dampers. If any one of the dampers is moved, or if the speed of one of the fans is changed, the pressures and flow rates in the system change everywhere. This cross-coupling complicates the control problem.

The third technical problem is that there are more control degrees of freedom than functional requirements. For example, in a variable-air-volume (VAV) air-handling unit with a supply and return fan, three control dampers, and a cooling coil, there are six control degrees of freedom. These six control elements must be operated to meet just three control functional requirements: adequate ventilation, appropriate building pressure, and adequate cooling. Since there are six control elements to meet three control requirements, there is often more than one way to meet the functional requirements at any given time. However, interacting and conflicting functional requirements and coupled control loops make the effective use of the extra control degrees of freedom a non-trivial problem.

## Project Objectives

The objectives of this project were as follows:

a. develop and demonstrate the benefit of a new, energy-efficient method of controlling air-handling equipment in buildings using optimization and computer simulation methods,
b. reduce fan power by 20%-40%,
c. reduce heating and cooling power by 10%, and
d. ensure that the controls are stable, and that minimum ventilation, appropriate building pressurization, and appropriate temperature control are achieved at all times.

## Project Approach

The project involved the development of a new, energy-efficient control strategy for air-handling systems. The new strategy was designed to achieve nearly optimal energy performance while meeting the functional requirements of the system. The development of the strategy made use of both mathematical models and optimization. However, the implementation of the strategy did not involve on-line optimization, nor the use of mathematical models. On-line optimization and model-based control strategies are usually not practical for controlling HVAC systems because of the expense and difficulty of obtaining sufficiently accurate process models. The resulting strategy is easy to implement and is computationally efficient.

Nearly all air-handling systems have more control degrees of freedom than control requirements. The extra control degrees of freedom offer the opportunity to improve the efficiency of the system. Whenever there is more than one way to meet the functional requirements of the system there will be just one way that minimizes power consumption. The proposed control strategy uses the extra control degrees of freedom to minimize power consumption while meeting the functional requirements of the system.

The approach involved first determining the optimal system behavior. An existing mathematical model of the system that included all of the pertinent information about fan characteristics, damper characteristics, and minor losses was used. The model was encoded with a commercially available computer simulation tool and tuned to match an air-handling system at the Elihu Harris State Office Building in Oakland, California. The model was combined with a commercially available optimization code to compute the optimal control settings (damper positions and fan speeds) that meet all functional requirements. The intent was to identify patterns of behavior that could be extracted from the optimal system behavior and encoded as rules or modes of operation for a finite state machine. Finite state machines are commonly used for process control, and have been proposed by Seem (1999) for controlling air-handling units.

We expected that the optimal behavior would involve sequential operation of the outdoor and return dampers. Typically these dampers are operated simultaneously. As one closes, the other opens. Exceptions are when plenum pressure control is being used to control minimum ventilation, and the strategy proposed by Seem et al. (2000) for volume matching systems. For the expected optimal behavior, the return damper should be completely open while the outdoor air damper modulates. When more outdoor air is needed, the outdoor air damper opens until it is completely open. If more is needed, then the return damper closes while the outdoor air damper remains completely open. Sequential operation should reduce the pressure drop and energy loss through the outdoor and return dampers. For example, when 50% outdoor air is needed, simultaneous operation would have the outdoor air damper half open and the return damper half open. Sequential operation would have both dampers wide open, so the pressure and energy loss through these two dampers would be almost eliminated at this outdoor air fraction.

We expected that the optimal behavior would also involve sequential operation of the exhaust damper and the return (or exhaust) fan if the system is VAV. If the building pressure needs to be reduced, then the exhaust dampers open while the return fan runs at a minimum speed until they are completely open. If the pressure is still too high, then the return fan speed is increased while the exhaust damper is held wide open. Control logic for responding to a reversed airflow condition would have the fan leading and the damper lagging. The operation of the return fan and exhaust damper could be based on building static pressure or a flow differential between the outdoor air intake and the exhaust flow rate.

We expected that sequential operation of the outdoor and return dampers and building pressure control or differential flow control with the exhaust damper would be nearly optimal for constant volume systems. For VAV systems, we expected that sequential operation of the outdoor and return dampers combined with sequential operation of the exhaust and return fan combined with static pressure reset control of the supply fan would be nearly optimal.

The basic idea behind the expected optimal behavior was that the controls should keep the control dampers as open as possible while ensuring that the control requirements are met. This is the idea behind static pressure resetting, where the supply fan is controlled so that the terminal dampers are as open as possible.

An advantage of the proposed strategy is that it is modular and decentralized. The economizer module that is best for a constant volume system is the same module that is best for a VAV system. For a VAV system, each module could be implemented by itself. The modules do not use information from the other modules.

The controls for each module would have dynamic interactions, but we expected that the interactions could be accommodated by controllers that are tuned so that they are slightly sluggish. This hypothesis was tested as part of this project.

The technical work was divided into seven basic tasks. The tasks are described below.

1. Identify a building system to model

We selected one of the air-handling units (AC-5) in the Elihu Harris State Office Building in Oakland, California. This system is a built-up VAV air-handling unit with a single supply fan and a single return fan. The design capacity of the system is 42,000 cfm of supply air. The air-handling unit serves three floors (floors 1-3) of the building totaling 45,800 square feet. There are 49 single-duct terminal units. Of the 49 terminal units, 30 have hot-water reheat coils. The controls operate as follows. The supply fan controls the static pressure in the second-floor supply duct near the junction with the main vertical supply shaft. The setpoint in use was 1.3 in. w.c. The three control dampers of the air-handling unit are electronically interlocked. Ventilation and building pressure are controlled with a volume-matching control strategy. The return fan is modulated to maintain a constant difference between the supply and return airflow rates. The setpoint for this difference was 8500 cfm. The controls use a temperature-based economizer that switches over when the return temperature exceeds the outdoor temperature. When the economizer is disabled, the outdoor air damper is moved to its minimum position, which is 15% open.

2. Adapt an existing model of air-handling systems to this application

We used the model described in Federspiel et al. (2001) to model AC-5. The model predicts velocities, pressures, and temperatures in air distribution networks. It predicts both transient and steady-state behavior.

The differential equations resulting from this model are "stiff", which means that there are some parts of the dynamics that are very fast relative to others. Solving stiff differential equations requires the use of a solver that is designed for stiff systems. We used a commercially available solver that uses a variable-order numerical differentiation formula (NDF). The NDF method is usually more efficient than Gear's method.

We did not model the dynamics of building heat transfer. Instead we used trend logs of the terminal flow rates of AC-5 recorded over several weeks to determine the the terminal flow rates required for temperature control. We used loss coefficients from ASHRAE (1993) for minor losses in the ductwork and for the characteristics of the air-handling unit control dampers. We used as-built drawings to determine dimensions of the ducts. We conducted an experiment on a VAV terminal unit to determine the characteristic of the VAV terminal dampers. We used manufacturer's data for the loss coefficients of the reheat coils and the characteristics of the fans. We used trend logs to tune the model. By tuning, we adjusted the loss coefficient of the supply duct filter and coil, and the loss coefficient of the return path so that the predicted supply and return flows matched the measured supply and return flows as closely as possible.

3. Determine optimal control settings

We used a commercially-available optimization tool to minimize power consumption while meeting the functional requirements for ventilation, building pressure, and temperature control. The optimizer uses a sequential quadratic programming method.

The optimization involved a search over a five-dimensional space (three damper positions and two fan speeds) with three inequality constraints and two equality constraints. The inequalities were as follows: 1) building pressure must be non-negative, 2) ventilation must meet or exceed code, 3) mixed-air temperature must meet or exceed a lower limit. The equality constraints were that the largest and smallest terminal flow errors must be nearly zero.

4. Develop control logic that will achieve nearly optimal control settings

We used the results of the optimization, our experience with systems of this type, trial and error, and results from Task 6 to design energy-efficient control strategies for VAV and constant volume air-handling systems. We discovered five facts about how operation of air-handling systems affects energy performance. We used these facts to design modes of operation and control logic that could be described by a hierarchical state chart.

5. Develop simple rules for setting the control loop gains

We developed a simple tuning rule similar to the tuning rule developed by Federspiel (1997) for the air-handling control loops. The tuning rule involves the use of integrating-only controllers. The tuning rule uses the design condition for the control loop, a gain margin, and a time constant. The same rule applies to all control loops except static pressure resetting based on VAV terminal position.

6. Develop simulation code for existing control strategies

We developed control logic to implement variants of two common, existing control strategies for VAV systems and the standard strategy for constant-volume systems. The strategies are as follows: 1) constant-volume with interlocked dampers, 2) VAV with volume matching, 3) VAV with plenum pressure control.

The first strategy involves constant-volume systems. We simulated a constant-volume system by adding the loss coefficient of a reheat coil to the branches of AC-5 that do not have a reheat coil, and running the fans at speeds that produced the design flow of the VAV system and neutral building pressure. The control dampers were interlocked.

Two variants of volume matching control were modeled: all three dampers interlocked, and the strategy developed by Seem et al. (2000), where the outdoor air damper is left wide open and the exhaust damper is interlocked with the return damper.

The plenum pressure control strategy studied by Sauer and Delp (1998) was also modeled. When the economizer is disabled, the outdoor airflow rate is controlled by setting the outdoor air damper to a fixed minimum position and using the return damper to regulate the mixed-air plenum pressure to a value that produces the desired flow through the outdoor air damper. The exhaust damper is modulated to control the building pressure.

7. Compare the new control strategy with commonly-used control strategies

We compared the energy performance of the existing strategies with the energy performance of the new strategy under four internal load conditions and five outdoor temperatures. Cooling power and fan power were predicted and compared.

## Project Outcomes

### Development of EASY

We developed and demonstrated the benefits of a new strategy that we call Energy-efficient Air-handling control StrategY (EASY). We used optimization and computer simulations to develop and demonstrate this new strategy.

<u>Modeling</u>

We used an existing model of air-handling systems for this project. The model initially used a fan component model similar to the fan component model used by other HVAC simulators such as HVACSIM+ (Park et al., 1986) and the simulation testbed of Haves and Norford (1998). This model uses two polynomial relationships, one between non-dimensional pressure and non-dimensional flow, and the other between non-dimensional power and non-dimensional flow. Mathematically, the pressure-flow-speed relationship is as follows:

$$C_P = \frac{P}{D^2 \Omega^2 r} \tag{1}$$

$$C_f = \frac{Q}{D^3 \Omega} \tag{2}$$

$$C_P = f_P\left(C_f\right) \tag{3}$$

where $f_P$ is a polynomial function, $P$ is the fan pressure, $D$ is the fan diameter, $\Omega$ is the fan speed, and $r$ is the density. The power-flow-speed relationship is as follows:

$$C_W = \frac{W}{D^5 \Omega^3 r} \tag{4}$$

$$C_W = f_W\left(C_f\right) \tag{5}$$

where $f_W$ is another polynomial function and $W$ is the fan power.

The fan model described by Equations 1-5 is usually used for design. The problem with using this model for simulating transient behavior is that the pressure, flow, and power coefficients all have the speed in the denominator, so the model has a singularity when the speed is zero. Simulators such as HVACSIM+ trap this condition and treat it as a special case.

For simulating transients, a speed of zero is not a special case. It is a common case. To avoid having extra logic for trapping this condition, we changed the fan model as follows. We defined the angle of attack of the fan as the angle between a tangent to the fan wheel and the velocity vector composed of a term due to rotation of the fan and a term due to flow through the fan. Mathematically, the model is as follows:

$$V_R = \frac{Q}{pD\Delta} \tag{6}$$

$$V_T = \frac{\Omega D}{2} \tag{7}$$

$$a = \sin^{-1}\left(\frac{V_R}{\sqrt{V_R^2 + V_T^2}}\right) \tag{8}$$

$$C_P^{'} = \frac{2P}{rV_m^2} \tag{9}$$

$$C_W^{'} = \frac{2W}{rV_m^3 pD\Delta} \tag{10}$$

$$C_P^{'} = f_P^{'}(a) \tag{11}$$

$$C_W^{'} = f_W^{'}(a) \tag{12}$$

where $V_R$ is the velocity normal to the fan wheel due to flow, $V_T$ is the velocity tangential to the fan wheel due to the rotation of the fan, $\Delta$ is the depth of the fan wheel, and $a$ is the angle of attack. The advantage of this model is that it only has a singularity when both the speed and flow of the fan are zero. Under this condition, the fan pressure and power are both zero. Another advantage is that when the speed is zero, the pressure coefficient is a standard loss coefficient, so it is easy to calibrate the model at the most extreme angles of attack.

<u>Optimization</u>

Each evaluation of the cost function (fan power plus cooling power) involved integrating a set of differential equations to steady-state. Since the differential equations were stiff, this was a computationally intensive process. We were able to increase the speed considerably by making the initial condition of the differential equation solver a global variable and resetting it every time the optimizer called that solver.

We found that the cost function contains many local minima. Figure 1 demonstrates this finding. The figure shows the power consumption at each iteration of the optimization for operation at high supply flow and high outdoor temperature. The optimizer converges to a total power consumption of 151.2 kW after 11 iterations. This power consumption is significantly higher than existing strategies operating under the same conditions. For example, the volume matching strategy with static pressure reset based on the critical damper position consumed 109.2 kW under this same condition.

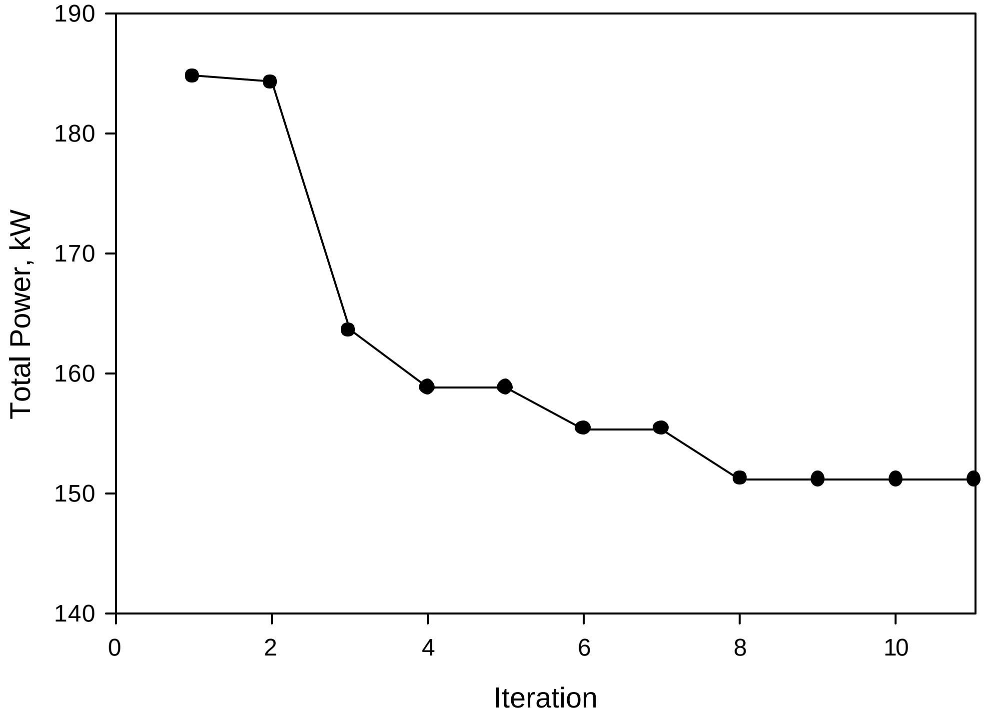


**Figure 1: Example of optimizer converging to a local minimum.**

After coding the existing strategies, we tried starting the optimizer at the operating condition of the best of the existing strategies at each load condition. We found that the optimizer could not improve the performance of the best existing strategy. This does not imply that the best existing strategy is globally optimal, only that it operates at a condition where there is a local minimum in the cost function.

The presence of many local minima made it difficult to use the optimizer as intended. We planned to derive control rules from globally optimal behavior. The local minima make it impossible to be certain that the results achieved by the optimizer are globally optimal. Since we could not be certain that optimization results were globally optimal, we relied primarily on trial and error, the performance and behavior of existing strategies, and experience to design efficient control strategies.

Design of EASY

From trial and error and the behavior of existing strategies, we learned the following facts that guided the design of EASY.

a. Reducing static pressure at part-load conditions has the biggest impact on energy performance.

Static pressure resetting has a 3.5 times bigger effect on reducing energy consumption than efficiently operating the air-handling unit control dampers and return fan. This is because the velocities through the air-handling unit dampers are low and the required return fan power is low. The required return fan power is low in part because the losses in the return path, which has a plenum return, are low.

b. The optimal static pressure is highly correlated with supply flow rate.

The supply flow rate is proportional to the cooling load, and the cooling loads from one zone to another are correlated.

c. Sequential operation of the dampers reduces fan power.

Sequential operation of the outdoor and exhaust dampers ensures that these dampers are as open as possible, reducing throttling losses through these dampers. This is most important when the outdoor temperature is low and the required supply airflow is high.

d. High building pressure reduces fan power when the economizer is enabled.

Under all conditions tested, neutral pressure required more fan power than positive pressure. This was due to a combination of low losses in the return path, low variable frequency drive (VFD) efficiency at low speed, and an oversized return fan. This combination is common. These three factors make it advantageous to use a low return fan speed. At 100% outdoor air conditions, the lowest fan power occurred when the building pressure was very high (> 1.0 in. w.c.). We selected an outdoor-exhaust flow differential equal to the minimum ventilation requirement so that building pressure was always positive, but not higher than some existing strategies make it already.

e. When the economizer is disabled, opening the outdoor air damper and return damper and closing the exhaust damper reduces fan power.

In this condition, envelope leakage is sufficient for exhausting outdoor air without generating excessively high building pressure. Opening the exhaust damper increases the pressures and flows required for the control loops to meet their setpoints.

Operating logic

EASY is designed to operate in different modes under different conditions. The modes of operation are both hierarchical and concurrent. The highest-level mode is economizer enabled or disabled.

The following is a description of the operating logic when the economizer is enabled. When the economizer is enabled, EASY operates the outdoor air dampers and return dampers sequentially rather than simultaneously. Together these two dampers control the supply air temperature. If the system were started with the outdoor air damper fully closed and the return damper fully open, EASY would first open the outdoor air damper to reduce the supply air temperature. If the supply air temperature were still too high when the outdoor air damper were fully open, then

EASY would close the return damper to lower the supply air temperature. The outdoor air damper and exhaust damper are interlocked when the economizer is enabled.

When the economizer is enabled, EASY uses the return fan to control the difference in the airflow rates through the outdoor air and exhaust dampers. The flow difference setpoint is equal to the minimum ventilation required by code.

EASY resets the supply duct static pressure. The preferred method of doing this is to reset based on supply flow. See the section on SAV for details.

When the economizer is disabled, the outdoor and return dampers are opened completely, and the exhaust damper is closed completely. The return fan is used to control the outdoor airflow rate to the minimum ventilation rate required by code. The supply duct static pressure is reset as when the economizer is enabled.

The operating logic and modes of operation can be described by the state chart shown in Figure 2. The state chart shows that in the economizer enabled mode there are two concurrent modes of operation: supply air temperature control and control of the difference between the outdoor airflow rate and the exhaust airflow rate. Also concurrent with these two control modes is the control of the supply fan which involves static pressure resetting.

When the outdoor air becomes hot enough, the economizer is disabled. In this mode the supply air temperature is controlled with the cooling coil (mechanical cooling). The dampers are moved to positions that conserve energy. The outdoor air and return dampers are opened, and the exhaust damper is closed. The flow control loop uses the return fan to control the outdoor airflow rate to the code requirement.

SAV

When static pressure is reset based on the critical damper position, the static pressure is correlated with supply airflow rate. Figure 7 shows the static pressure reset on critical damper position as a function of supply airflow rate for the volume matching strategy. The figure illustrates that the static pressure is lower at lower flow rates.

We used this finding to reset static pressure based on supply airflow. For the energy comparisons that follow, we chose a linear reset schedule. We configured the reset schedule so that at the design flow rate, the SAV reset schedule produced the same static pressure as resetting based on critical damper position. At no flow the reset schedule would produce a static pressure of 10 Pa.

This reset schedule does not produce optimal performance, but it is simple to implement in a retrofit application. In that case, the static pressure setpoint in place at the time would be used as the upper end of the reset schedule. There is some risk that the existing operating pressure is too high. It would be valuable to develop a commissioning method that could determine the best pressure for the top of the reset schedule, and also determine the best pressure for part-load conditions so that a nonlinear reset schedule could be used if appropriate.

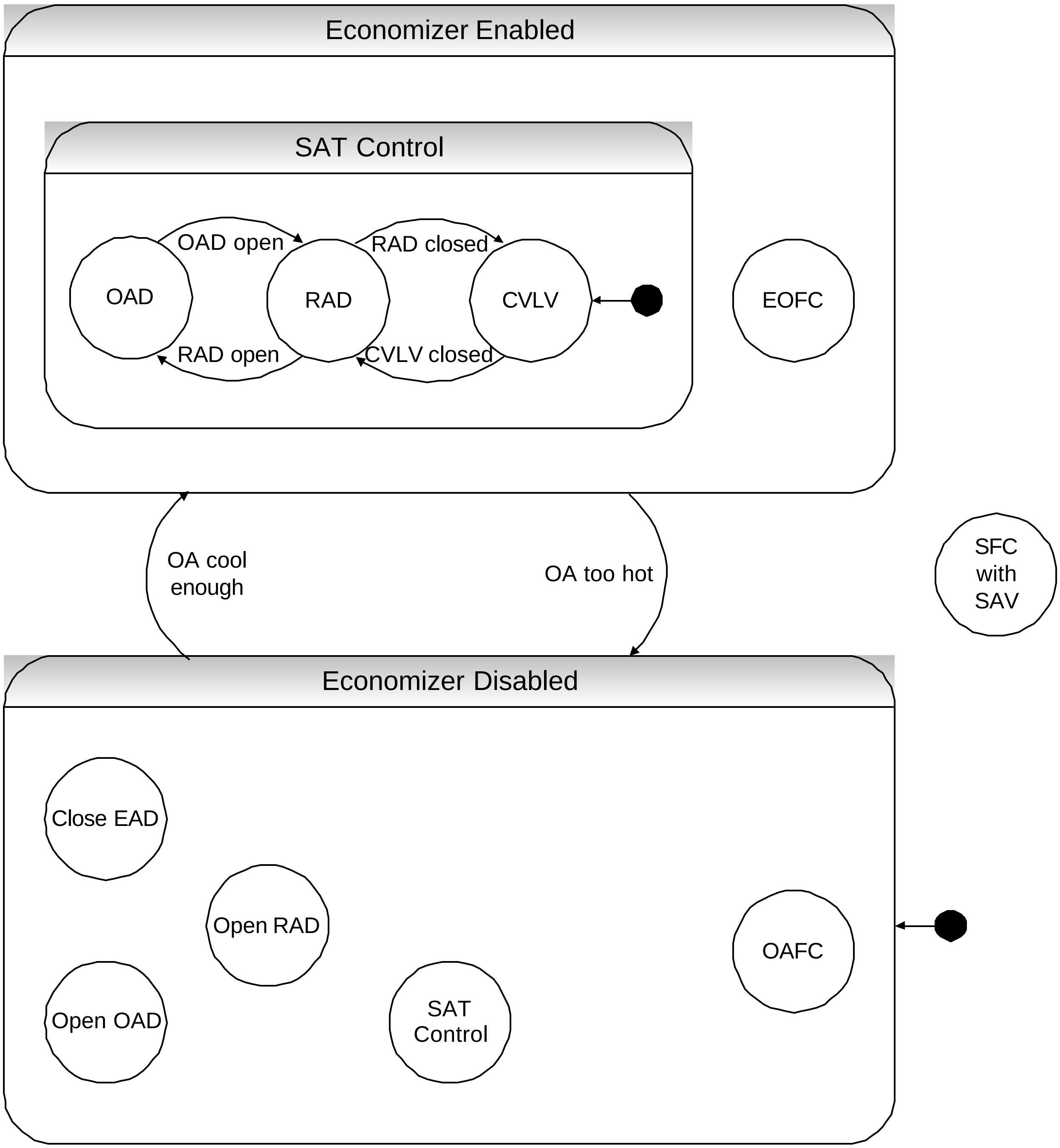


**Figure 2: State Chart of EASY**

## Energy performance

For VAV systems, we computed the energy performance of the three existing strategies and the EASY strategy for all combinations of five outdoor temperatures (40 – 80 °F in 10 °F increments) and four supply airflow rates (40% - 100% of design flow in 20% increments). We computed the performance without static pressure reset, with static pressure reset based on critical damper position, and with SAV. The average results are shown in Table 1. Detailed results are shown in the Appendix.

**Table 1: Average Power Consumption of VAV Systems (Watts per square foot).**

| | No resetting | | Optimized reset | | SAV | |
|---|---|---|---|---|---|---|
| | Fan | Cooling | Fan | Cooling | Fan | Cooling |
| EASY | 0.623 | 0.336 | 0.445 | 0.282 | 0.464 | 0.292 |
| Vol. Match | 0.635 | 0.337 | 0.454 | 0.282 | 0.474 | 0.292 |
| Seem | 0.636 | 0.336 | 0.455 | 0.282 | 0.474 | 0.292 |
| PPC | 0.676 | 0.407 | 0.491 | 0.305 | 0.481 | 0.294 |

The table shows that resetting the static pressure has a much bigger impact on energy performance than the way that the control dampers and return fan are controlled. The average difference in total power consumption between EASY and PPC is just 0.071 Watts per square foot. This is just an 8% improvement over PPC. The average difference between controls with critical damper resetting and no resetting was 0.248 Watts per square foot. This is a 25% reduction. The average difference between controls with SAV and no resetting was 0.231 Watts per square foot. This is a 23% reduction. SAV provides 93% of the benefit of critical damper resetting.

The results illustrate that the linear reset schedule of SAV is not optimal. At part-load, the linear reset chooses pressures that are higher than necessary. In this particular case, the linear resetting only reduces performance by 7% relative to critical damper resetting, but the relative difference between critical damper resetting and SAV could depend on the HVAC system design and load conditions. Additionally, we set the upper end of the SAV reset schedule to match the critical damper reset pressure because we knew it *a priori*. In practice it won't be known *a priori*. There is a need to develop a method to commission SAV so that the reset schedule is optimized for each specific system. This would enable additional savings and reduce the chance that SAV would starve some terminals.

Table 2 shows the energy performance of the standard method of controlling constant volume systems compared with EASY applied to constant volume systems. The table is the average power consumption at the same outdoor air temperatures used to produce Table 1. The detailed results are in the Appendix.

**Table 2: Energy Performance of Constant Volume Systems (Watts per square foot).**

| | Fan | Cooling |
|---|---|---|
| EASY | 1.193 | 0.489 |
| Standard method | 1.195 | 0.489 |

The table demonstrates that EASY offers little energy performance improvement over the standard method. EASY is slightly better at low temperatures, when it opens all three dampers substantially and reduces throttling losses slightly.

Sequential operation of the outdoor and return dampers produces less energy performance improvement than expected because the velocities through the control dampers are low, especially the velocity through the outdoor air damper.

**Control Performance**

<u>Transient Response</u>

We developed a tuning rule for local-loop controllers that is similar to the rule proposed by Federspiel (1997). The rule has three parameters, the static process gain, the time constant, and the gain margin. The rule uses a single-parameter controller that achieves zero steady-state error. This means integral-only control for systems that do not integrate or proportional-only control for systems that integrate. Dampers with floating actuators integrate, while dampers with pilot positioners do not integrate. Fans systems with variable frequency drives (VFD) also do not integrate VFDs have internal controls that keep the speed close to the speed command.

For systems that do not integrate, the tuning rule can be represented by the following differential equation:

$$\frac{d}{dt}C = \frac{1}{G}\frac{1}{M}\frac{1}{T}\left(S - V\right) \tag{13}$$

where C is the output of the controller, G is the process gain, M is the gain margin, T is the time constant, S is the setpoint, and V is the controlled variable. The process gain setpoint, and controlled variable must all have the same physical units. For all local loops except for static pressure reset based on critical damper position we used a gain margin of 5 and a time constant of 10 seconds. An example of this rule applied to the control of supply duct static pressure is as follows. The process gain used in Equation 1 is half the static pressure achieved by the supply fan at design flow. This value is readily available from design documents. It is approximately the highest supply duct static pressure that would be achievable when the fan is running at full speed. For AC-5 this quantity is 625 Pa. With the gain margin and time constant listed above and a setpoint of 325 Pa, the tuning rule for supply duct static pressure is as follows:

$$\frac{d}{dt}C = \frac{1}{625}\frac{1}{5}\frac{1}{10}\left(325 - P\right) \tag{14}$$

where C is now the command to the variable speed drive (a value between 0 and 1), and P is the supply duct static pressure in Pascals.

This tuning rule with a gain margin of 5 and a time constant of 10 worked for all local-loop controls except static pressure reset based on critical damper position. Table 3 shows the process gain used for all the local-loop controls studied.

**Table 3: Gains used for loop tuning.**

| Loop | Gain | Units |
|---|---|---|
| Supply duct static pressure | 1120 (4.5) | Pascals (in. w.c.) |
| Supply – return airflow rate | 17 (36000) | $m^3$/sec (cfm) |
| Outdoor airflow rate | 19.8 (42000) | $m^3$/sec (cfm) |
| Outdoor – exhaust airflow rate | 17 (36000) | $m^3$/sec (cfm) |
| Mixed-air plenum pressure | 625 (2.5) | Pascals (in. w.c.) |
| Return plenum pressure | 620 (2.5) | Pascals (in. w.c.) |
| Mixed-air temperature | 15 (27) | °C (°F) |

Figures 3-5 shows the startup transient response for EASY with SAV when the outdoor air temperature is 10 °C (50 °F) and the supply flow rate is 60% of design flow. The figure demonstrates that the tuning rule produces a settling time of approximately 30 minutes.

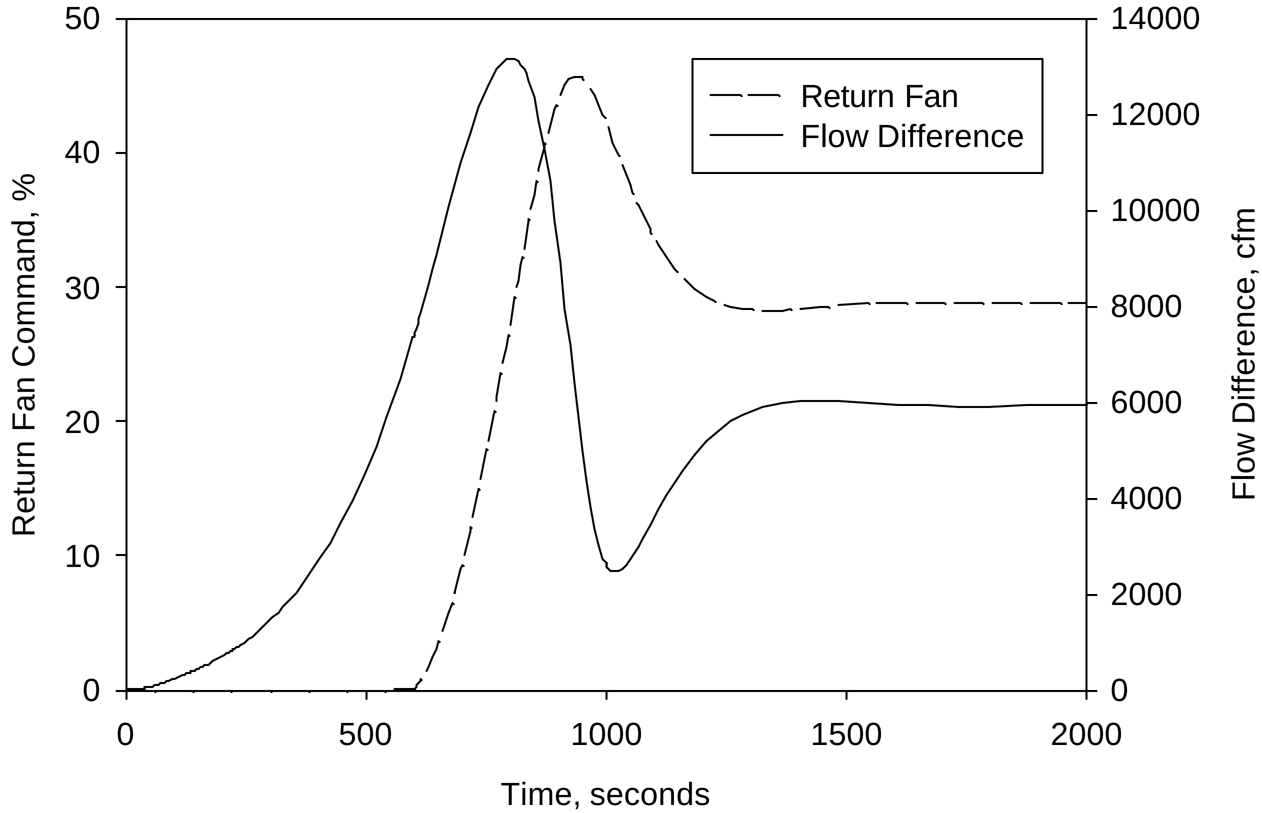


**Figure 3: Startup transient of EASY return fan loop.**

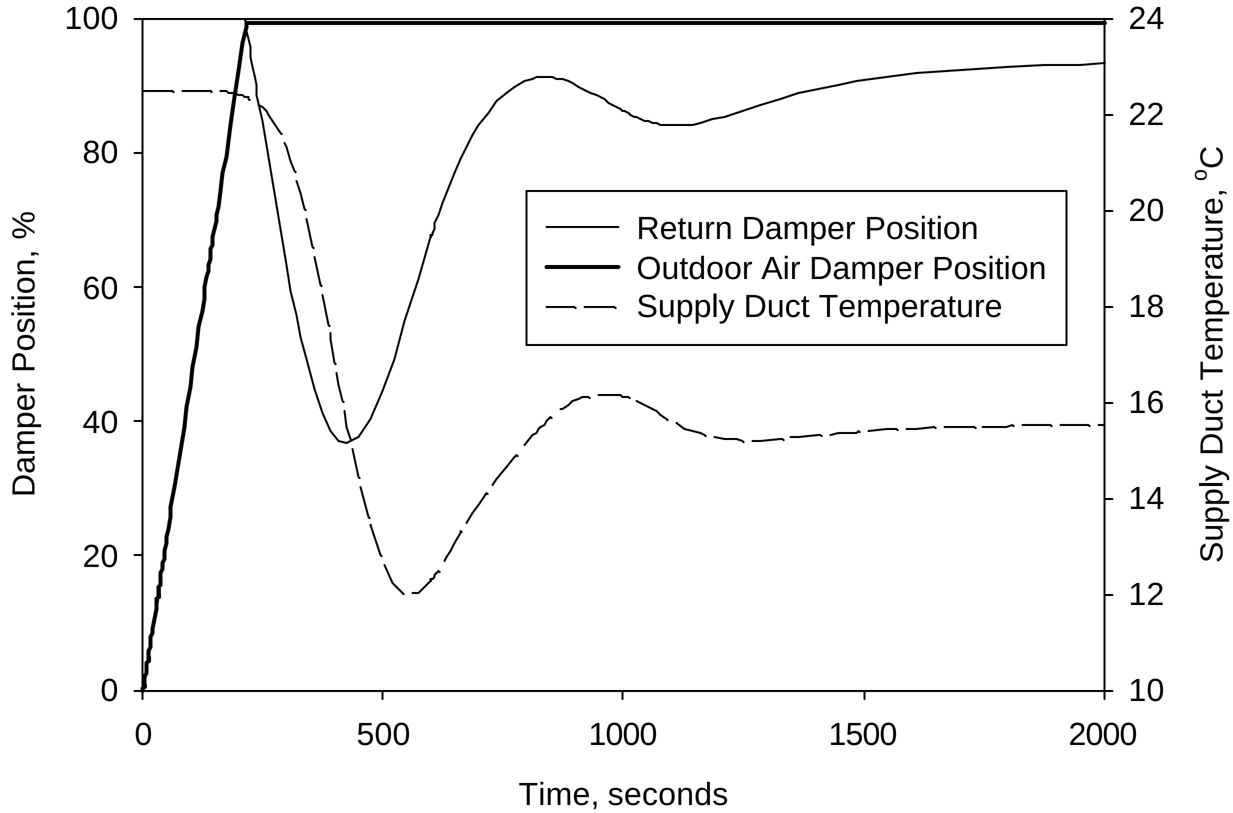


**Figure 4: Startup transient of EASY SAT loop.**

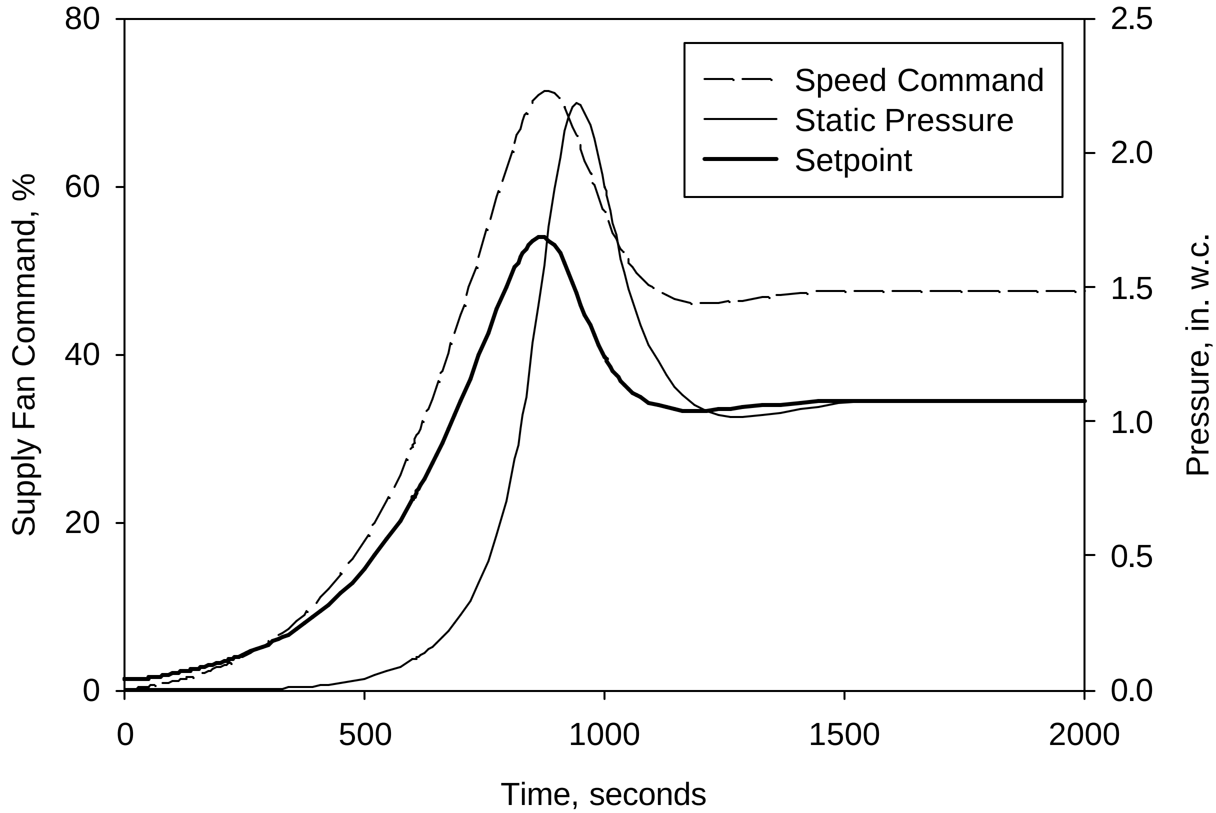


**Figure 5: Startup transient of EASY supply fan control loop.**

Static pressure reset based on critical damper position is a more complicated control loop than the others studied. The objective of static pressure reset is to minimize throttling losses in terminal dampers. In practice it is implemented in a variety of ways. To get the most energy savings from static pressure reset while ensuring that the terminals are always in control, the strategy must switch modes at the equilibrium point. When the most-open terminal is not completely open, then the controlled variable is position of that terminal and the controller will decrease pressure so that the most-open terminal will open more. When one or more terminals is completely open, then the controlled variable is the flow of the terminal that is completely open that is most below the setpoint. In this mode, the controller will increase the pressure so that flow will increase. The controls switch modes when just one damper is completely open and just in control, which is an equilibrium point if the control modes are both tuned properly.

We tried using the tuning rule described above to tune each mode of the static pressure reset controller. We found that this did not work. The gain of the damper position control loop was too high. This is because the critical damper position is not immediately responsive to fan speed changes. In comparison, the responses of the other controlled variables are immediately responsive to changes in actuation. For example, changes in outdoor air damper position immediately affect outdoor airflow rate. To stabilize the static pressure reset loop we had to reduce the gain by trial and error. Once stable, the response time was too slow to be useful in practice.

Figure 6 shows an example of the transient response of static pressure reset based on critical damper position as it is commonly implemented in practice. This simulation is for an outdoor temperature of 10 °C (50 °F) and a supply flow rate of 60% of design flow. The controller tries to regulate the average of the two most open dampers to a value of 90%. According to Taylor (2002), it is common practice to control the average of the terminal damper positions that exceed the 95th percentile. This prevents a failed damper from causing the fan controls to fail. For this simulation, the process gain was 10 (i.e., 10 times higher than would be used by the tuning rule). The response is very underdamped. The critical position oscillates significantly before reaching equilibrium. The settling time is nearly two hours.

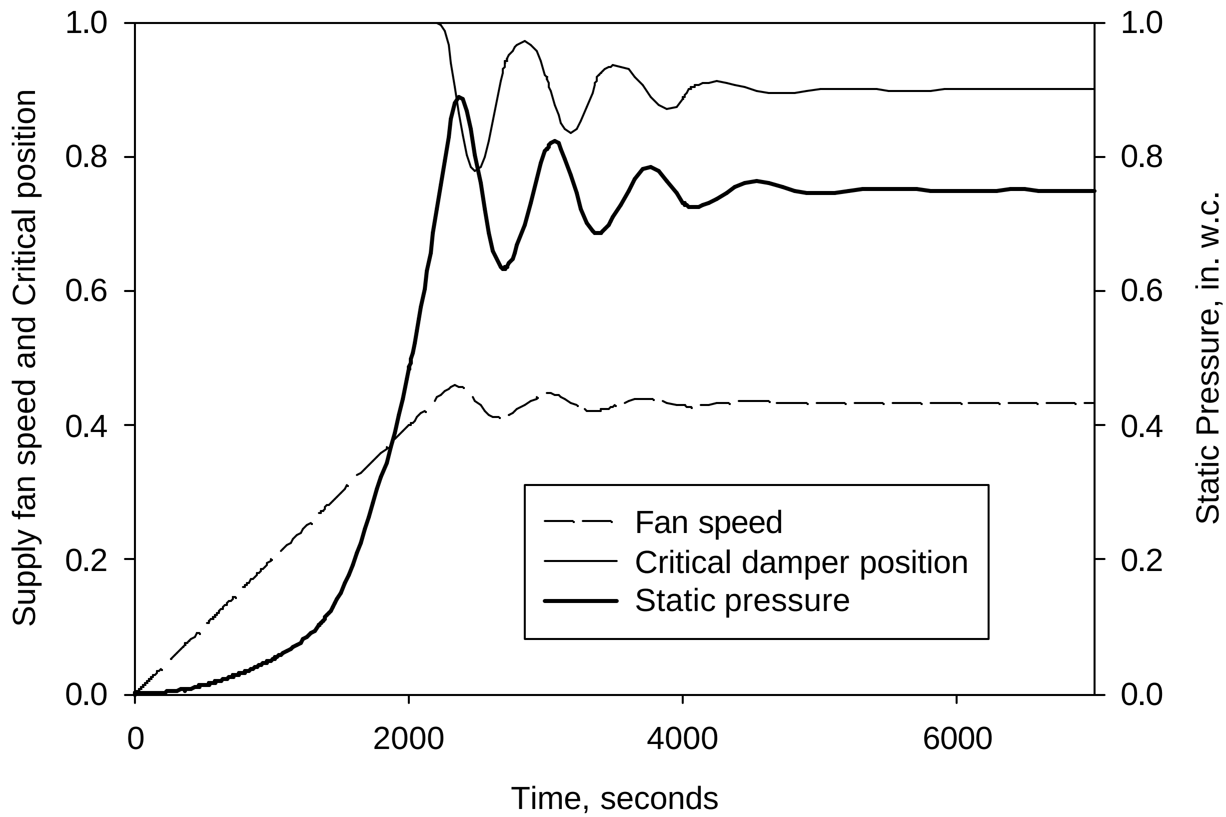


**Figure 6: Transient response of resetting on critical damper position.**

Meeting Control Objectives

We found that two of the existing control strategies did not meet control objectives under all operating conditions. The volume-matching strategy draws air into the exhaust duct when the economizer is disabled. This problem has been reported by Seem et al. (2000). The alternative volume matching control strategy that we tested is designed to solve that problem, and we found that it prevented flow reversal at the exhaust.

The PPC strategy caused excessively high building pressure (in excess of 1 in. w.c.) under the conditions of design flow and outdoor temperatures of 60 °F and 70 °F. The HVAC system was using 100% outdoor air at these temperatures. Under these conditions, the PPC strategy shut off the return fan because the return air plenum pressure was higher than the setpoint of 25 Pa even with the return fan off due to the high flow rate.

Both EASY and Seem's volume matching strategy ensured that all control objectives were met under all conditions. This did not necessarily make them more energy-efficient. When the PPC strategy allowed excessively high building pressures, it used less energy than any other strategy. However, closing the exhaust damper and keeping the outdoor and return dampers open when the economizer was disabled did make EASY and Seem's volume matching strategy slightly more efficient than the standard volume matching strategy and significantly more efficient that the PPC strategy.

## Conclusions

The conclusions of this project are as follows:

a. EASY reduces energy consumption by 2% - 10%.
b. EASY with SAV reduces fan power by 26.3% relative to a system without static pressure reset.
c. EASY with SAV reduces cooling power by 17.4% relative to a system without static pressure reset.
d. EASY and SAV are easier to tune that existing strategies, and they have a faster response time when tuned properly.
e. SAV can be implemented in the majority of existing VAV systems that do not have the capability of implementing static pressure reset based on terminal position.

## Recommendations

A field test is needed to quantify and validate the energy efficiency benefits. The tests should be conducted during the summer so that cooling energy benefits and fan energy benefits can be quantified.

Since most of the savings are derived from resetting static pressure, more research should be focused on static pressure resetting. Alternative strategies that can address the problems of resetting based critical damper position should be investigated. Methods for commissioning SAV so that it produces the largest energy benefit should be investigated.

## Public Benefits to California

California has not already benefited from this contract. However, the project demonstrated that the proposed technology could save 0.23W/ft$^2$/year in buildings with VAV air-handling units.

Based on figures from DOE (1998), we estimate that the proposed technology, if implemented in all applicable California buildings, would reduce energy costs by $171MM/year.

## Development Stage Assessment

Table 4 shows the development stages for the technology developed under this project. Supporting evidence for the completion ratings for each activity are show below.

Marketing

The targeted market segment is buildings with VAV air-handling systems. According to DOE (1998), VAV system serve 13.5 billion square feet of space in the U.S. The California market is approximately one-seventh of the U.S. market. The average number of annual operating hours is 4150 (DOE, 1998). The technology is able to reduce average consumption by 0.23 Watts per square foot. Nationwide, the average cost of electricity in 2000 was $0.072/kW-h. In California it was $0.093/kW-h. Combining these figures, the technology could save $928MM/year nationwide and $171MM/year in California. The savings can be achieved either by programming existing programmable control systems or by installing a stand-alone programmable control system.

Engineering/Technical

The existing technical gap is whether or not the computer-based predictions will hold up in the field. The concept is a control strategy involving a combination of hardware and software. In many retrofit situations, the concept would only involve software. The approach is to change the way that the control loops work. The technology is simpler, more reliable, easier to commission, and uses less energy than competing control strategies. We set performance goals of 20%-40% reduction in fan power and 10% reduction in cooling power. The project showed that the technology reduces fan power by 26.3% and reduces cooling power by 17.4%. We are working on a pilot field test with a potential customer. It will be conducted in a building in Sacramento during the summer so that fan energy and cooling energy benefits can be demonstrated.

Legal/Contractual

A patent search indicated that SAV is patentable. An application was filed with the USPTO.

RiskAssess/Quality Plans

There are no anticipated environmental risks. The pilot will be conducted in a building where a failure of the pilot will not be a critical problem.

Strategic

This project is linked to the PIER Buildings Issue 1 and Issue 4. This project does not impact other projects. It is not critically dependent on any other projects.

Production Readiness

The top candidates for commercialization partners are large organizations that could benefit from the energy savings, and manufacturers who could embed the technology in their products and sell it as a value-added feature. Partners of both kinds have been identified.

Public Benefits/Cost

The technical feasibility test did not change the public benefit to cost ratio. The benefits still outweigh the costs.

**Table 4: Development Assessment Matrix**

| Stages / Activity | 1 Idea Generation | 2 Technical & Market Analysis | 3 Research | 4 Technology Develop-ment | 5 Product Develop-ment | 6 Demon-stration | 7 Market Transfor-mation | 8 Commer-cialization |
|---|---|---|---|---|---|---|---|---|
| **Marketing** | | | | | | | | |
| **Engineering / Technical** | | | | | | | | |
| **Legal/ Contractual** | | | | | | | | |
| **Risk Assess/ Quality Plans** | | | | | | | | |
| **Strategic** | | | | | | | | |
| **Production. Readiness/** | | | | | | | | |
| **Public Benefits/ Cost** | | | | | | | | |

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## Glossary

AHU: Air-Handling Unit
CV: Constant Volume
CVLV: Cooling Valve
DDC: Direct Digital Control
EAD: Exhaust Air Damper
EOFC: Exhaust minus Outdoor air Flow Control
EASY: Efficient Air-handling StrategY
HVAC: Heating, Ventilating, and Air-Conditioning
NDF: Numerical Differentiation Formula
OA: Outdoor Air
OAD: Outdoor Air Damper
OAFC: Outdoor Air Flow Control
PPC: Plenum Pressure Control
RAD: Return Air Damper
SAT: Supply Air Temperature
SAV: Static pressure Adjustment from Volume flow
SFC: Supply Fan Control
SPR: Static Pressure Reset
SPRCDP: Static Pressure Reset based on Critical Damper Position
SVM: Seem's Volume Matching strategy
VAV: Variable-Air-Volume
VFD: Variable Frequency Drive

## Appendix

The following tables show the energy performance results of individual computer simulations of VAV systems.

**Table 5: Fan Power (W/ft$^2$) for EASY without SPR.**

| | Fraction of design flow, % | | | |
|---|---|---|---|---|
| OAT (°F) | 40 | 60 | 80 | 100 |
| 40 | 0.255961 | 0.376092 | 0.638668 | 1.211354 |
| 50 | 0.255502 | 0.375633 | 0.638515 | 1.213297 |
| 60 | 0.256485 | 0.37762 | 0.642751 | 1.222227 |
| 70 | 0.256485 | 0.37762 | 0.642751 | 1.222227 |
| 80 | 0.256026 | 0.376943 | 0.641703 | 1.220393 |

**Table 6: Cooling Power (W/ft$^2$) for EASY without SPR.**

| | Fraction of design flow, % | | | |
|---|---|---|---|---|
| OAT (°F) | 40 | 60 | 80 | 100 |
| 40 | 0 | 0 | 0 | 0 |
| 50 | 0 | 0 | 0 | 0 |
| 60 | 0.054803 | 0.077445 | 0.124607 | 0.223057 |
| 70 | 0.438275 | 0.544607 | 0.713341 | 0.977817 |
| 80 | 0.604476 | 0.736026 | 0.946987 | 1.283341 |

**Table 7: Fan Power (W/ft$^2$) for EASY with SPRCDP.**

| | Fraction of design flow, % | | | |
|---|---|---|---|---|
| OAT (°F) | 40 | 60 | 80 | 100 |
| 40 | 0.038013 | 0.152249 | 0.460721 | 1.12059 |
| 50 | 0.037729 | 0.151747 | 0.460328 | 1.122314 |
| 60 | 0.037838 | 0.152664 | 0.463821 | 1.129694 |
| 70 | 0.037817 | 0.152664 | 0.463712 | 1.13083 |
| 80 | 0.037707 | 0.152118 | 0.462926 | 1.128362 |

**Table 8: Cooling Power (W/ft$^2$) for EASY with SPRCDP.**

| | Fraction of design flow, % | | | |
|---|---|---|---|---|
| OAT (°F) | 40 | 60 | 80 | 100 |
| 40 | 0 | 0 | 0 | 0 |
| 50 | 0 | 0 | 0 | 0 |
| 60 | 0.008624 | 0.031834 | 0.089803 | 0.205742 |
| 70 | 0.25321 | 0.411965 | 0.636201 | 0.947162 |
| 80 | 0.382205 | 0.577031 | 0.854345 | 1.245742 |

**Table 9: Fan Power (W/ft$^2$) for EASY with SAV.**

| | Fraction of design flow, % | | | |
|---|---|---|---|---|
| OAT (°F) | 40 | 60 | 80 | 100 |
| 40 | 0.082096 | 0.186703 | 0.457948 | 1.121179 |
| 50 | 0.081812 | 0.186201 | 0.457445 | 1.122664 |
| 60 | 0.082118 | 0.187314 | 0.46083 | 1.131179 |
| 70 | 0.082118 | 0.187314 | 0.46083 | 1.131201 |
| 80 | 0.081921 | 0.186921 | 0.46 | 1.129563 |

**Table 10: Cooling Power (W/ft$^2$) for EASY with SAV.**

| | Fraction of design flow, % | | | |
|---|---|---|---|---|
| OAT (°F) | 40 | 60 | 80 | 100 |
| 40 | 0 | 0 | 0 | 0 |
| 50 | 0 | 0 | 0 | 0 |
| 60 | 0.018144 | 0.038865 | 0.089214 | 0.206026 |
| 70 | 0.311201 | 0.436878 | 0.634847 | 0.947293 |
| 80 | 0.45262 | 0.607293 | 0.852686 | 1.246245 |

**Table 11: Fan Power (W/ft$^2$) for Volume Matching without SPR.**

| | Fraction of design flow, % | | | |
|---|---|---|---|---|
| OAT (°F) | 40 | 60 | 80 | 100 |
| 40 | 0.261397 | 0.387096 | 0.66441 | 1.275109 |
| 50 | 0.261943 | 0.388341 | 0.666223 | 1.273996 |
| 60 | 0.256485 | 0.37762 | 0.642751 | 1.222162 |
| 70 | 0.256485 | 0.37762 | 0.642751 | 1.222162 |
| 80 | 0.261943 | 0.383035 | 0.648144 | 1.227969 |

**Table 12: Cooling Power (W/ft$^2$) for Volume Matching without SPR.**

| | Fraction of design flow, % | | | |
|---|---|---|---|---|
| OAT (°F) | 40 | 60 | 80 | 100 |
| 40 | 0 | 0 | 0 | 0 |
| 50 | 0 | 0 | 0 | 0 |
| 60 | 0.054803 | 0.077445 | 0.12460699 | 0.223035 |
| 70 | 0.438275 | 0.544607 | 0.71334061 | 0.977817 |
| 80 | 0.605786 | 0.73738 | 0.94844978 | 1.285066 |

**Table 13: Fan Power (W/ft$^2$) for Volume Matching with SPRCDP.**

| | Fraction of design flow, % | | | |
|---|---|---|---|---|
| OAT (°F) | 40 | 60 | 80 | 100 |
| 40 | 0.039039 | 0.156769 | 0.479716 | 1.180197 |
| 50 | 0.03869 | 0.157533 | 0.481878 | 1.17952 |
| 60 | 0.037795 | 0.152773 | 0.463712 | 1.131004 |
| 70 | 0.037838 | 0.152598 | 0.463777 | 1.130611 |
| 80 | 0.042052 | 0.156703 | 0.468559 | 1.1369 |

**Table 14: Cooling Power (W/ft$^2$) for Volume Matching with SPRCDP.**

| | Fraction of design flow, % | | | |
|---|---|---|---|---|
| OAT (°F) | 40 | 60 | 80 | 100 |
| 40 | 0 | 0 | 0 | 0 |
| 50 | 0 | 0 | 0 | 0 |
| 60 | 0.008624 | 0.031921 | 0.08978166 | 0.205983 |
| 70 | 0.253275 | 0.411921 | 0.63624454 | 0.947096 |
| 80 | 0.382271 | 0.57821 | 0.85563319 | 1.247969 |

**Table 15: Fan Power (W/ft$^2$) for Volume Matching with SAV.**

| | Fraction of design flow, % | | | |
|---|---|---|---|---|
| OAT (°F) | 40 | 60 | 80 | 100 |
| 40 | 0.083996 | 0.192249 | 0.476681 | 1.18048 |
| 50 | 0.084127 | 0.193166 | 0.478843 | 1.180371 |
| 60 | 0.082118 | 0.187314 | 0.46083 | 1.131179 |
| 70 | 0.082118 | 0.187314 | 0.46083 | 1.131179 |
| 80 | 0.086419 | 0.191638 | 0.465568 | 1.136681 |

**Table 16: Cooling Power (W/ft$^2$) for Volume Matching with SAV.**

| | Fraction of design flow, % | | | |
|---|---|---|---|---|
| OAT (°F) | 40 | 60 | 80 | 100 |
| 40 | 0 | 0 | 0 | 0 |
| 50 | 0 | 0 | 0 | 0 |
| 60 | 0.018144 | 0.038865 | 0.08921397 | 0.206026 |
| 70 | 0.311201 | 0.436878 | 0.63484716 | 0.947271 |
| 80 | 0.453624 | 0.608341 | 0.85395197 | 1.24786 |

**Table 17: Fan Power (W/ft$^2$) for Seem's Strategy without SPR.**

| | Fraction of design flow, % | | | |
|---|---|---|---|---|
| OAT (°F) | 40 | 60 | 80 | 100 |
| 40 | 0.260022 | 0.387358 | 0.669825 | 1.294236 |
| 50 | 0.261616 | 0.389563 | 0.671594 | 1.288188 |
| 60 | 0.256485 | 0.37762 | 0.642751 | 1.222162 |
| 70 | 0.256485 | 0.37762 | 0.642751 | 1.222162 |
| 80 | 0.256026 | 0.3769 | 0.641463 | 1.21976 |

**Table 18: Cooling Power (W/ft$^2$) for Seem's Strategy without SPR.**

| | Fraction of design flow, % | | | |
|---|---|---|---|---|
| OAT (°F) | 40 | 60 | 80 | 100 |
| 40 | 0 | 0 | 0 | 0 |
| 50 | 0 | 0 | 0 | 0 |
| 60 | 0.054803 | 0.077445 | 0.124607 | 0.223035 |
| 70 | 0.438275 | 0.544607 | 0.713341 | 0.977817 |
| 80 | 0.604694 | 0.73631 | 0.947336 | 1.283712 |

**Table 19: Fan Power (W/ft$^2$) for Seem's Strategy with SPRCDP.**

| | Fraction of design flow, % | | | |
|---|---|---|---|---|
| OAT (°F) | 40 | 60 | 80 | 100 |
| 40 | 0.037773 | 0.155633 | 0.48321 | 1.197817 |
| 50 | 0.037948 | 0.157533 | 0.486004 | 1.193297 |
| 60 | 0.037795 | 0.152642 | 0.463952 | 1.130546 |
| 70 | 0.037838 | 0.152686 | 0.463668 | 1.131419 |
| 80 | 0.037686 | 0.15238 | 0.462686 | 1.127555 |

**Table 20: Cooling Power (W/ft$^2$) for Seem's Strategy with SPRCDP.**

| | Fraction of design flow, % | | | |
|---|---|---|---|---|
| OAT (°F) | 40 | 60 | 80 | 100 |
| 40 | 0 | 0 | 0 | 0 |
| 50 | 0 | 0 | 0 | 0 |
| 60 | 0.008624 | 0.031834 | 0.089825 | 0.205895 |
| 70 | 0.253231 | 0.411965 | 0.636179 | 0.947402 |
| 80 | 0.382205 | 0.577511 | 0.854629 | 1.246048 |

**Table 21: Fan Power (W/ft$^2$) for Seem's Strategy with SAV.**

| | Fraction of design flow, % | | | |
|---|---|---|---|---|
| OAT (°F) | 40 | 60 | 80 | 100 |
| 40 | 0.082598 | 0.191441 | 0.480109 | 1.198319 |
| 50 | 0.083493 | 0.193406 | 0.482926 | 1.194083 |
| 60 | 0.082118 | 0.187314 | 0.460939 | 1.131179 |
| 70 | 0.082118 | 0.187314 | 0.460939 | 1.131201 |
| 80 | 0.081943 | 0.1869 | 0.459825 | 1.128908 |

**Table 22: Cooling Power (W/ft$^2$) for Seem's Strategy with SAV.**

| | Fraction of design flow, % | | | |
|---|---|---|---|---|
| OAT (°F) | 40 | 60 | 80 | 100 |
| 40 | 0 | 0 | 0 | 0 |
| 50 | 0 | 0 | 0 | 0 |
| 60 | 0.018144 | 0.038865 | 0.089214 | 0.206026 |
| 70 | 0.311201 | 0.436878 | 0.634847 | 0.947293 |
| 80 | 0.452795 | 0.607533 | 0.853013 | 1.246616 |

**Table 23: Fan Power (W/ft$^2$) for PPC without SPR.**

| | Fraction of design flow, % | | | |
|---|---|---|---|---|
| OAT (°F) | 40 | 60 | 80 | 100 |
| 40 | 0.265066 | 0.395371 | 0.678079 | 1.290022 |
| 50 | 0.265087 | 0.395175 | 0.676921 | 1.285066 |
| 60 | 0.26238 | 0.389607 | 0.664825 | 1.172009 |
| 70 | 0.26238 | 0.389607 | 0.664825 | 1.172009 |
| 80 | 0.278275 | 0.407358 | 0.685218 | 1.282991 |

**Table 24: Cooling Power (W/ft$^2$) for PPC without SPR.**

| | Fraction of design flow, % | | | |
|---|---|---|---|---|
| OAT (°F) | 40 | 60 | 80 | 100 |
| 40 | 0 | 0 | 0 | 0 |
| 50 | 0 | 0 | 0 | 0 |
| 60 | 0.052991 | 0.752642 | 0.121834 | 0.236594 |
| 70 | 0.436201 | 0.542227 | 0.71048 | 0.991441 |
| 80 | 0.608799 | 0.741747 | 0.954803 | 1.29393 |

**Table 25: Fan Power ($W/ft^2$) for PPC with SPRCDP.**

| | Fraction of design flow, % | | | |
|---|---|---|---|---|
| OAT (°F) | 40 | 60 | 80 | 100 |
| 40 | 0.038755 | 0.161878 | 0.49214 | 1.195699 |
| 50 | 0.038755 | 0.162009 | 0.492052 | 1.191201 |
| 60 | 0.038428 | 0.159913 | 0.482882 | 1.087402 |
| 70 | 0.038537 | 0.159847 | 0.482882 | 1.087249 |
| 80 | 0.046856 | 0.172882 | 0.501332 | 1.188341 |

**Table 26: Cooling Power ($W/ft^2$) for PPC with SPRCDP.**

| | Fraction of design flow, % | | | |
|---|---|---|---|---|
| OAT (°F) | 40 | 60 | 80 | 100 |
| 40 | 0 | 0 | 0 | 0 |
| 50 | 0 | 0 | 0 | 0 |
| 60 | 0.008035 | 0.030415 | 0.087358 | 0.216223 |
| 70 | 0.252271 | 0.410218 | 0.63369 | 0.957358 |
| 80 | 0.384258 | 0.581026 | 0.861332 | 1.255721 |

**Table 27: Fan Power ($W/ft^2$) for PPC with SAV.**

| | Fraction of design flow, % | | | |
|---|---|---|---|---|
| OAT (°F) | 40 | 60 | 80 | 100 |
| 40 | 0.085109 | 0.198188 | 0.489389 | 1.19583 |
| 50 | 0.085218 | 0.198253 | 0.488974 | 1.191594 |
| 60 | 0.084454 | 0.195459 | 0.479869 | 1.088144 |
| 70 | 0.084454 | 0.195459 | 0.479869 | 1.088144 |
| 80 | 0.095328 | 0.209629 | 0.498297 | 1.190415 |

**Table 28: Cooling Power ($W/ft^2$) for PPC with SAV.**

| | Fraction of design flow, % | | | |
|---|---|---|---|---|
| OAT (°F) | 40 | 60 | 80 | 100 |
| 40 | 0 | 0 | 0 | 0 |
| 50 | 0 | 0 | 0 | 0 |
| 60 | 0.017162 | 0.037271 | 0.08679 | 0.216397 |
| 70 | 0.309847 | 0.435044 | 0.632314 | 0.957729 |
| 80 | 0.455328 | 0.611659 | 0.859651 | 1.256572 |

The following tables show the energy performance results of individual computer simulations of constant volume systems.

**Table 29: Fan and Cooling Power (W/ft$^2$) of EASY for CV system.**

| OAT (°F) | Fan | Cooling |
|---|---|---|
| 40 | 1.19231441 | 0 |
| 50 | 1.192641921 | 0 |
| 60 | 1.193799127 | 0.199847162 |
| 70 | 1.193799127 | 0.961768559 |
| 80 | 1.192358079 | 1.281179039 |

**Table 30: Fan and Cooling Power (W/ft$^2$) of Standard CV system.**

| OAT (°F) | Fan | Cooling |
|---|---|---|
| 40 | 1.196200873 | 0 |
| 50 | 1.196703057 | 0 |
| 60 | 1.193799127 | 0.20139738 |
| 70 | 1.193799127 | 0.963318777 |
| 80 | 1.192401747 | 1.2819869 |

This following figure shows the static pressure reset based on critical damper position as a function of supply airflow for the volume matching strategy.

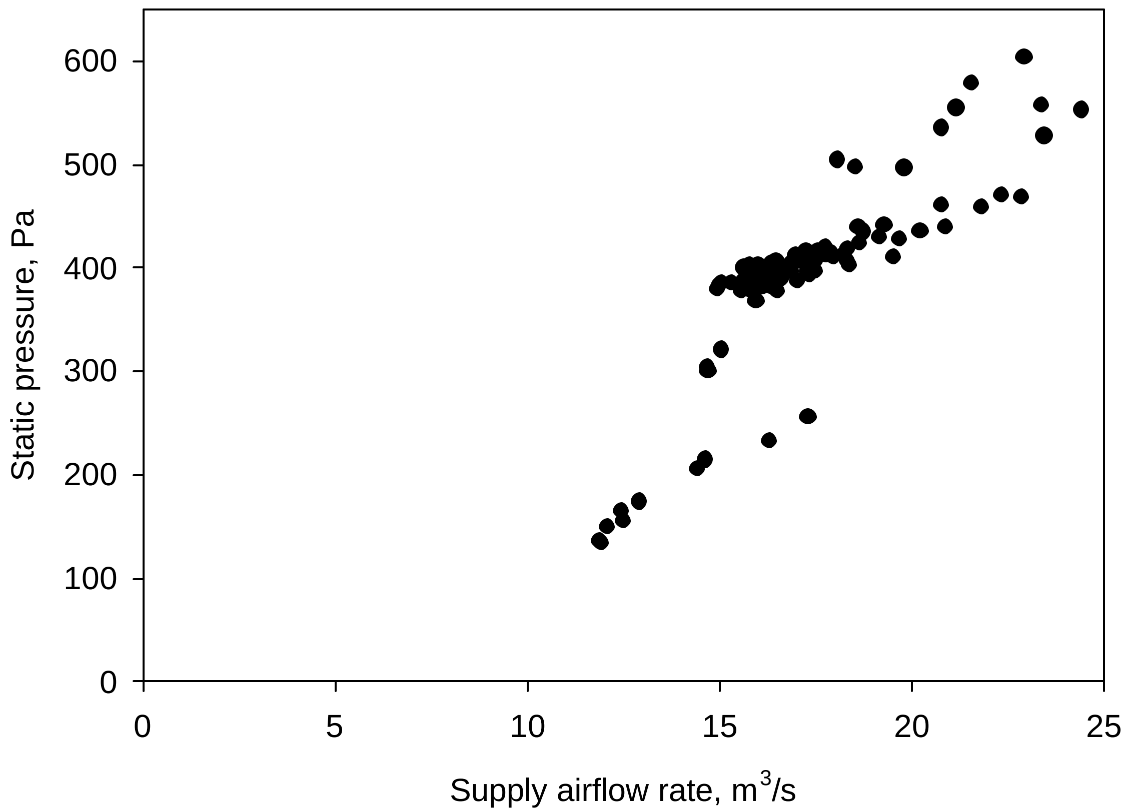

Figure 7: Reset static pressure versus supply airflow rate for VM strategy.